TABBED TRANSFER FINS AND AIR-COOLED HEAT EXCHANGERS WITH TABBED FINS

Contractural Origin of the Invention

The United States Government has rights in this invention under Contract No. DE-AC36-99GO-10337 between the United States Department of Energy and the National Renewable Energy Laboratory, a Division of the Midwest Research Institute.

Cross Reference to Related Applications

This application claims the benefit of U.S. Provisional Application No. 60/486,071, filed July 10, 2003, which is incorporated by reference herein in its entirety.

Background of the Invention

Field of the Invention.

The present invention relates generally to heat exchangers that utilize fins or plates on or in contact with tubes, pipes, or plates to transfer heat away from the working fluid in the tubes, pipes, or plates, and more particularly, to heat transfer fins, and heat exchangers or condensers that include such fins, that include a plurality of tabs extending from the fins to provide enhanced heat transfer on the air side of the heat exchanger with low and acceptable increases in pressure drop.

Relevant Background.

Heat exchangers are used extensively in industrial and consumer applications, and typically employ two moving fluids, one fluid being hotter than the other, to transfer heat to the colder fluid. Many heat exchangers currently in use, such as in air conditioners, automotive radiators, process industry air-cooled condensers, and boilers, transfer heat between a gas and a single or multi-phase liquid. Typically, such heat

exchangers include a number of liquid conduits, e.g., circular, oval, or flat tubes, pipes, or plates, that are positioned within a shell or housing that defines a gas flow passage or chamber. The heat exchanger uses a fan or blower to force a gas, e.g., air, to flow within the gas flow chamber in a perpendicular (i.e., cross-flow) or parallel (i.e., counter-flow) direction relative to the liquid conduits. The resulting heat transfer between the liquid and the gas is directly proportional to the heat transfer surface area between the liquid and the gas, to the temperature difference between the liquid and the gas, and to the overall heat transfer coefficient of the heat exchanger. The overall heat transfer coefficient is defined in terms of the total thermal resistance to heat transfer between the gas and the liquid, and it is dependent on a number of characteristics of the heat exchanger design, such as the thermal conductivity of the material used to fabricate the conduit and the local film coefficients along the conduit, i.e., measurements of how readily heat can be exchanged between the gas and the exterior surfaces of the conduit.

Although gas-liquid heat exchangers are widely used, the heat transfer effectiveness of these heat exchangers is low. The low heat transfer effectiveness leads to relatively high operating and capital costs for gas-liquid heat exchangers because a greater number of units and/or larger capacity units that require more power must be used to obtain a desired heat transfer. For example, air-cooled geothermal power plants operate at low temperature differences between the gas and the liquid and, in these power plants, more than 25 percent of the cost of producing electricity is the expense of purchasing and operating gas-liquid heat exchangers or condensers. As a result of these high costs, continuing efforts are being made to improve heat transfer effectiveness of gas-liquid heat exchangers while at the same time controlling the manufacturing and operating cost to increase the likelihood that new heat exchanger designs will be adopted by industry and consumers.

Geothermal plants provide one example of a situation in which there is often not a sufficient supply of water or other cooling liquid for evaporative cooling, and heat

must be rejected to atmospheric air. This heat rejection is accomplished through the use of large air-cooled condenser units in which air is forced through several rows of long individually finned tubes by large fans, i.e., a gas-liquid heat exchanger or condenser is employed. Each of the tubes carrying the hot working fluid has fins on their outer surfaces in order to provide a large heat transfer surface area. Finned-tube heat exchangers have been used for many years to improve the gas-side heat transfer rate by increasing the heat transfer surface area available for contacting the gas as it flows through the heat exchanger. In general, finned-tube heat exchangers are cross-flow heat exchangers that include a number of tubes, i.e., conduits, for carrying the liquid fabricated from aluminum, copper, steel, or other high thermal conductivity materials. The tubes pass through and contact a series of parallel, high thermal conductivity material sheets or plates, i.e., fins, which provide an extended heat transfer area for the tubes. The overall heat transfer area is based on the number and size of the included fins. The fins are separated a fixed distance, i.e., a fin separation distance, and define relatively parallel channels that direct the gas flow across and among the tubes. Heat transfer occurs as the gas flows through the channel and contacts the surface of the fins and as the gas contacts the outer surfaces of the tubes. The highest heat transfer rate on a flat surface like a flat fin occurs at the leading edge of the surface and decreases with distance from the leading edge as a boundary layer develops and thickens causing the local heat transfer coefficient to decrease.

However, although finned-tube heat exchangers are widely used because they are relatively inexpensive to produce and do not create a large pressure drop, there are several operational drawbacks to finned-tube heat exchangers. For example, finned-tube heat exchangers have low heat transfer coefficients on large portions of the fins due to the development of thick boundary layers. Additionally, these heat exchangers have poor heat transfer in the wake or shadowed regions behind tubes as a majority of the gas

flowing over a tube does not contact the back side of the tube or contact the portion of the fin surface that is shadowed by the tube.

In an attempt to increase the effectiveness of finned-tube heat exchangers, efforts have been made to vary the surface and overall geometry of the parallel fins to interrupt gas boundary layers or to make it more difficult for thick boundary layers to form on the fins. For example, finned-tube heat exchangers have utilized triangular or s-shaped wavy fins to enhance the heat transfer coefficient by disrupting boundary layer development and, also, by increasing the available heat transfer area. Alternatively, the surface geometry of flat, parallel fins can be enhanced, as is often done in refrigerant condensers, by slitting the fin three or four times in the areas of the fin between the tubes, thereby interfering with boundary layer development by creating offset surfaces on the fin that cause repeated growth and wake destruction of boundary layers. A number of heat exchangers have been developed that include structures on the fin surfaces that are designed to create turbulence in the channel between the fins to break up the boundary layer and increase heat transfer. Generally, these structures have been configured with a major portion of their surface area, such as winglets, vortex generators, and the like, facing the flowing gas or directed toward or into the gas flow path, e.g., to have a large profile relative to the gas flow path within the fin channel. However, the larger the profile or "form" placed in the flow path of the gas, including the liquid tubes, the larger the pressure drop in the cooling gas as form drag is increased, which is generally an undesirable and often unacceptable result.

While some of the above changes in the fin surface and fin shape may provide somewhat higher heat transfer coefficients in finned-tube heat exchangers, the design changes also result in unacceptably large increases in pressure drop on the gas side of the heat exchanger that require increased expenditures on fan power. Additionally, many of these design changes have not been adopted due to unacceptably high manufacturing costs in producing the fins or due to increased maintenance costs as some

of the fin surface structures snag or collect debris often found in unfiltered air often used in air-cooled heat exchangers.

Hence, there remains a need for a more effective finned-tube, gas-liquid heat exchanger that provides improved heat transfer capabilities on the gas side of the exchanger while creating an acceptable increase in the pressure drop for the gas passing through the tubes and fins and while controlling manufacturing and maintenance costs.

Disclosure of the Invention

The present invention addresses the above problems by providing an improved design for heat transfer fins that enhances the heat transfer rate on the gas or air side of heat exchangers with relatively low increase in pressure drop. Briefly, the fins include numerous tabs or secondary fins that are bent upward and downward from the body of the fin at a selected bend angle (such as between about 70 and 110 degrees and more typically, about 90 degrees). In this manner, the material of the fin body is retained for use in heat transfer with the air or gas flowing over the fins. Preferably, all or a majority of the tabs are aligned with the flow path(s) of the cooling gas to minimize the creation of turbulence and pressure drop (i.e., by minimizing creation of flow drag by only "showing" the tab's leading edge to the flowing gas).

For example, the tabs may be substantially planar and aligned with their surfaces parallel to the main flow path or simple flow path or line (or in some cases, the local flow paths) of the cooling gas relative to the fin. In a first embodiment, the tabs are positioned with their planar surfaces perpendicular to a leading edge of the fin to align the tabs substantially parallel with the main flow path of gas across the fin. In a second embodiment, some or all of the tabs are positioned to be more aligned with local flow paths or with streamlines to guide air flowing in the channel between fins to reduce the size of wakes behind tubes and to reduce pressure drop relative to the first embodiment

by producing less turbulent flow. In the second embodiment, the tabs may be positioned substantially parallel to or angled less than about 5 degrees relative to the streamlines. This is achieved by positioning the tabs at various, differing offset angles, e.g., 0 degrees (or substantially parallel to the simple flow path), 10 degrees as measured from either side of the simple flow path, and the like. The offset angles are typically less than about 20 degrees and more preferably less than about 10 degrees as measured from either side of the simple flow path with the offset angle, at least in some embodiments, being selected to be substantially parallel (such as within 5 to 10 degrees or less to being parallel) to the local stream line or flow path. In this manner, heat transfer is significantly enhanced by reducing the thickness of the thermal boundary layer on each tab and by placing heat transfer surface area in contact with cooler portions of the flowing gas (for a cooling application), e.g., the surface area of the tabs extends outward into cooler portions of the flow channel between adjacent fins.

The tabs of the fin serve four main functions. First, the tabs are preferably arranged so as to serve as a plurality of sites for starting new boundary layers. This is achieved generally by offsetting the tabs (or adjacent rows of the tabs) such that downstream tabs are not shadowed by upstream tabs. Second, the tabs are preferably positioned relative to the flowing gas to enhance heat transfer. More particularly, the tabs typically have a tab height as measured from the surface of the fin body that allows the tab to extend out into the region of high air flow rate and cool air (in the case of cooling applications), i.e., forming on both sides of the fin body. In one embodiment, the fin height is selected to between about 40 and 50 percent (e.g., about one half) of the size of the channel between adjacent fins, i.e., a fin separation distance and tabs are extended outward from both sides of the fin body. In other embodiments, the fin height is greater than 50 percent with one specific embodiment using a tab height of about two thirds or about 67 percent of the fin separation distance. In this manner, the tabs place fin material into the coolest portion of the gas flowing on both sides of the tab. Third,

the openings in the main fin surface disrupt the boundary layer on that surface thus enhancing heat transfer. Fourth, due to their angles, their flow resistance, and the channels they create, the tabs direct air flow so that the fin surface is more uniformly covered and relatively stagnant wake regions behind tubes are reduced. To achieve these functions, the tabs are formed by punching holes in the fin body but retaining a connection to the fin body on at least one edge. The material is then bent upward and/or downward relative to the fin body to extend at a bend angle from one or both of the surfaces of the fin body, i.e., to allow the tabs to extend into the boundary layers that form on one or both sides of a fin.

According to one aspect of the invention, a method is provided for fabricating heat transfer fins for heat exchangers. The method comprises providing a plain fin, such as an aluminum fin typically utilized in finned-tube heat exchangers. A tab pattern is selected or provided for the particular fin to define the quantity, size, and location of heat transfer tabs on the fin. The tab pattern selection may comprise performing a variety of flow and heat transfer tests on the fin implementing a number of potential tab patterns to obtain a useful pattern to enhance heat transfer while not unacceptably increasing pressure drop. With a tab pattern selected, a punch mechanism or tool can be fabricated or provided based on the pattern. The punch mechanism can be adapted for punching the tabs in one operation with tabs extending from one or both sides of the fin body. The method continues with forming, such as with the punch mechanism, the heat transfer tabs defined by the tab pattern by creating openings or holes in the fin by removing material from the fin body while retaining a connecting edge between the fin body and the removed material or tab body. The forming comprises bending the removed fin body material along the connector edge to a bend angle, such as 90 degrees, relative to one of the two sides of the fin body. The tab pattern is configured such that all or a majority of the tab bodies are aligned parallel (or within about 10 to 20 degrees) to a simple flow path (i.e., a directional line drawn perpendicular to the leading edge of

the fin body) or are aligned parallel (or within about 5 to 10 degrees) of local flow paths. In one embodiment, the tab pattern is configured such that the surface area of the removed material or tab bodies is such that the tabbed fin has porosity of less than about 50 percent and more typically between about 15 and 30 percent. In some embodiments, about half of the tabs extend from one side of the fin body while the remaining tabs extend from the second side of the fin body. Preferably, the tabs on each side of the fin body are arranged in the tab pattern such that adjacent upstream and downstream tabs (or proximal and distal tabs relative to a fin body leading edge) are offset to avoid shadowing of downstream tabs. The tabs are also arranged in such a way that they do not adversely interfere with the tabs on adjacent fins. Further, their pattern encourages uniform flow over the main fin and maximized heat conduction within the fin.

Brief Description of the Drawings

Figure 1 is a simplified heat exchanger according to the present invention illustrating one configuration in which tabbed fins or plates (such as those shown in later figures) can be employed to enhance heat transfer on the air or gas side;

Figure 2 illustrates two fins according to the invention, one that is partially punched or has less tab density and one that is fully fabricated or has higher tab density, and a template that can be used for producing a tool to fabricate the fins shown;

Figure 3 is a partial cross section of a set of fins (prior to placement on tubes) illustrating a side view of the fins, i.e., a view showing the planar surface of the tabs extending outward in this embodiment from both sides of the fins, in accordance with the present invention;

Figure 4 is a partial sectional view of a heat exchanger similar to that shown in Fig. 3 illustrating more clearly the mating of the fins to a liquid tube and one exemplary

configuration for the tabbed fins of the present invention (with fewer fins shown than would typically be included for ease of illustration);

Figure 5 is another partial sectional view of a pair of fins according to the invention illustrating selection of the height of the tabs to reach the region of high air flow and coolest air (in a cooling application);

Figure 6 is a top view of a portion of a fin fabricated according to the present invention which is useful for illustrating that the tabs in adjacent rows are offset from each other to enhance heat transfer and for illustrating one useful pattern of the tabs on a fin relative to the fin collars/tube openings or to the tubes in a heat exchanger assembled according to the invention;

Figure 7 is a cross sectional view of the fin shown in Fig. 6 taken at line 7-7 illustrating one arrangement for tabs on a fin (e.g., extending from both sides or surfaces of the fin body) and showing the bend angle and height of the tabs, in accordance with the present invention;

Figure 8 is a perspective view of another fin fabricated according to the invention and further illustrating the concept of offset tabs utilized in most fin embodiments;

Figure 9 is a flow diagram for air or gas relative to a fin, such as the fin shown in Fig. 6 that can be utilized for designing the tab pattern on a fin, such as for selecting the alignment of the tabs relative to the leading edge of the fin body, in accordance with the present invention;

Figure 10 is a perspective view of a portion of another embodiment of a tabbed fin according to the invention showing the use of tabs positioned at offset angles to align

the tabs substantially parallel with local flow paths or streamlines and to redirect flow to reduce the size of wakes and distribute the flow more uniformly over the fin surface;

Figure 11 is a view of one of the heat transfer surfaces or sides of the tabbed fin of Fig. 10 provided to better illustrate the use of a number of offset angles to align all or most of the fins with local flow paths and to illustrate the use of rectangular tabs (e.g., shown by the holes created by the removal of material from the fin body to form the tab bodies), in accordance with the present invention;

Figure 12 is an illustration of flow within a channel adjacent to a fin without tabs as modeled with a bubbler device showing typical path lines and showing typical wakes formed behind heat transfer tubes, in accordance with the present invention;

Figure 13 is an illustration of the invention similar to Fig. 12 showing modeled flow within a channel adjacent a tabbed fin according to the invention having rectangular tabs aligned parallel to the simple flow path and providing reduced wakes behind tubes;

Figure 14 is another illustration of the invention similar to Fig. 12 showing modeled flow within a channel adjacent another tabbed fin according to the invention again having square tabs but with at least a portion of the tabs positioned at small offset angles relative to the simple flow path to direct flow and to create reduced wakes compared to the fin of Fig. 13;

Figures 15-17 illustrate graphically the tested performance of one embodiment of tabbed fins of the present invention compared with performance of plain fins both in an 8 fin per inch (FPI) configuration and with performance of the tabbed fin design as indicated by a set of ratios commonly used for heat transfer design to compare a new design with an older or original design; and

Figures 18-20 illustrate graphically the tested performance of the tabbed fin of Figures 16 and 17 in a 10 FPI configuration again comparing performance of the tabbed fin to a plain fin and providing values of heat transfer design ratios achieved with the tabbed fin, in accordance with the present invention.

Detailed Description of the Preferred Embodiments

The present invention is directed to heat exchanger fins employing tabs to increase heat transfer effectiveness, heat exchangers or condensers incorporating such tabbed fins or plates such as air-cooled, finned-tube heat exchangers, and methods of making tabbed fins. Generally, each fin of the invention includes a multiplicity of small tabs or secondary fins that are formed by punching material (i.e., metal) out of the main fin body (i.e., creating a hole or opening) and the punched material is bent outward away from the main fin body in one or both directions from the surface of the fin. To minimize or control the creation of a vortex or increased pressure drop, the tabs are generally planar and aligned with the direction of the fluid, e.g., air, flowing over the fins in the channel between adjacent fins. In other words, a leading edge of the tab first contacts the flowing gas and the substantially planar body of the tab is aligned substantially parallel to the gas flow path or direction over the fin.

In some embodiments, it is assumed that there is one flow direction through the fins, such as perpendicular to the leading edge of the fins, and all of the fins are aligned parallel to this flow direction. These embodiments can be described as tabbed fins with tabs positioned at a zero offset angle relative to the simple flow path (i.e., a line drawn perpendicular to the leading edge of the fin body) such that the planar tab bodies are substantially parallel to the simple flow path. In other embodiments, two or more flow directions within the fin channel are identified, and fin tabs in different locations of the fin are aligned with these different flow paths to better limit creation of pressure drop. The different flow paths can be termed "local flow paths" or "local streamlines" and in

these embodiments, a portion or all of the tabs are positioned at offset angles greater than zero relative to the simple flow path (such as less than 20 degrees and more preferably less than about 10 degrees offset). In some cases, a portion or subset of the tabs are aligned somewhat, such as less than about 5 percent or about 5 degrees, off of the local flow direction or local flow path in order to redirect flow into areas of low flow such as the area behind or shadowed by a tube, whereby heat transfer is enhanced without creating a vortex at the tab.

The tabs can take many shapes, such as square, rectangular, triangular, semi-circular, and a combination of these shapes, and are generally bent outward at about a right angle relative to the body of the fin but a smaller bend angle can be utilized to practice the invention. The tabs can also be curved, e.g., into an L-shape or U-shape, to place more tab area into the most advantageous flow regions. Tabs in adjacent rows are preferably offset from each other so as to avoid shadowing of subsequent tabs and to promote the creation of multiple boundary layers within the channels. The tabs have a height defined by the amount of material removed from the fin body, and this tab height is generally (but not necessarily, such as when a bend angle of less than 90 degrees is utilized) less than the separation distance between fins, i.e., fin separation distance, and in many embodiments, the tab height is selected to be about one half and three fourths of the fin separation distance, e.g., two thirds or 67 percent, to place a large portion of surface area of the tab bodies within the coolest air flowing between the fins, i.e., at about the top of the boundary layer formed by the fin body. As will become clear from the following description, the use of tabbed fins according to the invention can significantly enhance the air-side heat transfer coefficient in finned-tube heat exchangers, with some tests indicating an increase of up to approximately 100 percent with a corresponding increase in pressure drop of approximately 60 percent relative to smooth or plain fins. Additional test results are provided and explained with reference to Figures 15-20 that, briefly, indicate significant increases in heat transfer coefficients

for tabbed fins according to the invention over plain fins with smaller corresponding increases in pressure drop.

In the following description, the use of tabbed fins according to the invention is explained most fully with finned-tube arrangements in which the fins or plates are arranged in a parallel fashion. However, it should be understood that the invention covers the use of tabs on many other arrangements of fins than just the ones shown. For example, it is anticipated that tabbed fins would be useful with helically wound fins. Additional applications of tabs to fins, whether or not the fins are applied to tubes, will be understood by those skilled in the art and are considered within the breadth of the invention and following description.

Figure 1 illustrates a simplified heat exchanger 100 that may be configured with tabbed fins according to the present invention. The heat exchanger 100 is a finned-tube heat exchanger or condenser and is shown in simplified form for ease of description. Generally, the heat exchanger 100 would further include a housing enclosing fins and tubes and, at least in part, defining air flow channels, i.e., causing the air or other gas to flow between the fins and to define a gas inlet and a gas outlet. The heat exchanger 100 further would include one or more fans to draw (or push) air across the tubes and between the fins. These components are well known and hence, it is not believed necessary to illustrate or describe these components further to allow one skilled in the arts to understand and practice the invention. The heat exchanger 100 transfers heat energy from one fluid, i.e., the fluid in, F_{IN}, to another fluid or gas, i.e., the air in, A_{IN}, which results in a cooler fluid being output, i.e., the fluid out, F_{OUT}, and a hotter fluid or gas being output, i.e., the air out, A_{OUT}, from the heat exchanger 100. Of course, the fluids being cooled may be a gas or liquid or any mixture thereof, and the invention applies to heating as well as cooling.

Referring again to Figure 1, the heat exchanger 100 includes a plurality of plates or fins 110 that are arranged in a parallel manner and separated a fin separation distance. The fins 110 are tabbed, not shown in Figure 1 for simplicity's sake but described fully in the following paragraphs and shown in detail in subsequent figures, to enhance heat transfer on the gas or air side of the exchanger 100. The fins 110 are typically pressed or fit together and spaced apart a fin separation distance by a set of tube collars 115 provided on one (or both) sides of each tube opening in the fin 110. The tube collars 115 are sized to receive the liquid tubes 120, 130, such as copper, steel, or other metal tubes or pipes, which may have a circular cross section or another cross sectional shape such as oval, flat, rectangular, and the like (with the size, shape, location, material, and other properties of the tube not being limited to the invention). The collars 115 provide the heat transfer surface between the tubes 120, 130 and the fins 110 and are often press fit onto the tubes, such as by inserting the tubes 120, 130 and then over pressurizing the tubes 120, 130 to cause the tubes 120, 130 to expand and contact the collars 115. Of course, fins may be attached to tubes in other ways including welding, brazing and the like and winding or wrapping (as is the case in foil fins that are helically wound onto tubes).

The incoming air, A_{IN}, is passed through the channels between the fins 110 and strikes the leading edges of the plurality of secondary fins or tabs (which are shown in a rows that are diagonally offset in Figures 2-9 but other patterns can be used such as those in Figures 10 and 11), thereby significantly increasing the amount of heat transferred from the incoming liquid, F_{IN}, via the tubes 120, 130, to the outgoing air, A_{OUT}. Additionally, tabs on the fins 110 typically are all substantially perpendicular to the leading edges 140. The tabs, or at least a portion of the tabs, can also be arranged or aligned at different angles, i.e., offset angles, chosen such that during operation of the heat exchanger 100 the tabs direct flow smoothly around the tubes 120, 130 (and collars 115). In this manner, the tabs on the fins 110 can be utilized to shrink the wakes behind

the tubes 120, 130. It should be further noted that although tubes 120, 130 are shown in Figure 1, the tabbed fin concept of the present invention could also be used in other heat exchangers, including those that do not use tubes. For example, but not as a limitation, tabs similar to the ones described herein may be used on serpentine-like fins in plate heat exchangers.

Referring now to Figure 2, portions of fins (or test fin sections) manufactured according to the present invention are shown along with a template useful in creating a punch tool for forming the holes and tabs in the fin bodies. The fin sections shown were formed for ½-inch tubing but, of course, the size of the tube may be varied to practice the invention (e.g., it is anticipated that the tabbed fins will be useful with 1-inch tubing that is often used in heat exchangers). The fin 210 is formed from a metal (such as aluminum, copper, or other metal useful on the air or gas side of a heat exchanger) sheet that is fully manufactured or punched with numerous tabs extending outward from both sides or surfaces of the fin body, such as with about 50 percent extending upward and about 50 percent extending downward. Although it may not be clear from Figure 2, the tabs remain attached to the body of the fin 210, 220 along a seam or connecting portion, i.e., to retain the heat transfer material and surface area of the original fin body, and are bent outward so when the fin 210, 220 is used in a heat exchanger the tabs extend into the air stream flowing over the fin 210, 220 (as is explained in detail below). The tabs thus act as secondary fins or secondary fin surfaces.

As shown, a relatively large portion of the fin body surface area has been removed to form the tabs. For example, the surface area of the fin body removed or used to form the tabs may be selected from the range of 0 to 50 percent, and more preferably between 10 and 40 percent, and in one preferred embodiment, the surface area removed is about 20 to 25 percent of the fin body surface area, i.e., 20 to 25 percent of the original fin body surface area or material is used to form the tabs. While initially it may appear that the area used to form the tabs should be maximized, there are

limitations to how much material can or should be removed from the fin body. More particularly, the removal of fin body material reduces the volume of or mass of the fin body that is available to conduct the heat from the tube collar and tube contacting the collar to the tabs. Hence, testing may likely be required to identify for a particular fin and tube arrangement the amount of fin body surface area that should be used in forming the tabs. The amount of material removed defines the surface porosity of the fins 210, 220, and this porosity may be varied to practice the invention and may vary with fin materials and the makeup of the fluids passed through the heat exchanger. In one embodiment, the fin porosity is selected to be between 20 and 30 percent with tested embodiments utilizing about 25 percent porosity. The number and size of the tabs may be varied while maintaining a desired porosity with larger tabs resulting in fewer tabs and smaller tabs resulting in fins with more tabs. Also, the amounts of heat transfer improvement and pressure drop increase can be controlled by varying the tab dimensions (height and length) and the number of tabs or porosity.

The tabs shown are generally square (as can be seen clearly from the material removed to form the tab openings or holes) but numerous other shapes can be utilized, such as rectangular (such as shown in Figures 10 and 11), triangular, trapezoidal (e.g., with 0 to 10 degree or more angled leading and trailing edges) and semi-circular. The tabs can also be bent into shapes such as L-bends and U-bends. As will be explained below, the size of each tab is typically dictated by its height, i.e., what distance the tab is to extend away from the fin body, and its length. For example, if a tab is square and the height is selected to be one half of the distance between adjacent fins, then the sides of the square tab would each have a length equal to the tab height or one half the fin separation distance. The tabs are shown to be arranged in relatively linear rows in fin 210 and are shown to be more dense in areas of the fin in which flow is expected to be higher or highest, e.g., between tubes, in front of tubes, and away from housing wall surfaces. The fin 220 illustrates the use of a much smaller percentage of the fin body

surface area to form tabs, and is also useful for showing an embodiment of the invention in which the fins are not necessarily arranged in neatly linear rows but, instead, "downstream" (e.g., tabs more distal from a leading edge of the fin body) tabs are offset. In other words, downstream tabs are generally not positioned immediately behind an upstream tab or in the same air flow path to minimize shadowing and to encourage development of new boundary layers by each tab. Item 230 is a template useful for designing a punch tool for creating the tabs in a plain fin with darkened or colored holes indicating tabs to be punched to extend from a first surface of the fin body and the other holes indicating tabs punched the other direction. Again, the template 230 is useful for showing that downstream tabs are offset from upstream tabs. Arrangement of the tabs is also done with due consideration for providing heat conduction paths in the base fin from the tube to outer regions of the fin.

Figure 3 illustrates a sectional view of a series of fins formed according to the present invention. As shown, air, A_{IN}, enters the channel formed between adjacent fin bodies and the heated air, A_{OUT}, exits the other end of the fin channel or air flow chamber of the heat exchanger. The view of Figure 3 can be thought of as a side view and shows that in this embodiment the fins include a plurality of tabs that extend outward from the fin body on both sides. As shown, the tabs are relatively planar with a square cross section. The planar portion of the tabs (i.e., the larger surface area portion of the tabs) are bent away from the fin body to be substantially parallel to the direction of the air, A_{IN}, through the channel between adjacent fin bodies or substantially parallel to the simple flow path in the channel. The fins or fin bodies are separated by a distance (typically determined by the height of the tube collars) that is shown as $D_{FIN\ SEPARATION}$ or $D_{F.S.}$. In the illustrated embodiment, the tabs have a height (as measured from the fin body surface to the distal edge of the tab) that is less than about half of the $D_{FIN\ SEPARATION}$. In this manner, the tabs in one fin do not typically contact tabs in adjacent fins (which may cause assembly problems) but can be positioned within the cooler

portion of the air, A_{IN}, such as at the top of the boundary layer (distal to the fin body surface). In other embodiments (not shown), the tabs extend from only one side of the fin body, and in some embodiments (such as those shown in Figures 10 and 11), the fin tabs extend outward more than one half of the fin separation, $D_{FIN\ SEPARATION}$, to a distance of about two thirds of the $D_{FIN\ SEPARATION}$. Another embodiment is to have fins with tabs punched out of one side of the fin only and with these fins arranged back-to-back such that the combined fin has everywhere conduction paths for heat transfer (i.e., no openings completely through the fins exist). It is also possible to combine these tabs with some tabs that reach all the way across the gap to serve as spacers. In addition, some of the tabs can be twisted to introduce vorticity.

Figure 4 illustrates in more detail a sectional view of a heat exchanger of the present invention. As shown, a pair of fin bodies 420 are placed in contact with each other via tube collars 422 and the collars are press-fit onto a tube 430 that is used for carrying a hot fluid, F_{IN} through the heat exchanger. The pair of fins are separated by a distance, $D_{F.S.}$, and the tabs 404, 410 shown on the fin bodies 420 extend outward from both surfaces 424, 428 of the fin bodies 420 a tab height, H_{TAB}. As discussed previously, the tab height (or distance at which they extend when not configured with a bend angle of approximately 90 degrees relative to the body surface 424, 428) is typically selected to be less than the distance separating the fins, $D_{F.S.}$, and more typically, as shown, is selected to be about 40 to 50 percent of this distance, $D_{F.S.}$. In other embodiments, the tab height, H_{TAB}, is greater than 50 percent of the distance, $D_{F.S.}$, to place more surface area of the tabs 404, 410 into the cooler portion of the air flow (rather than just the tip of the tabs 404, 410), and in one embodiment, the tab height, H_{TAB}, is selected to be between about 50 and 75 percent and more preferably about two thirds or about 67 percent of the distance, $D_{F.S.}$.

The tab 404 is shown to include a leading edge 406 and a trailing edge 408. The tab 404 is bent or formed in a manner that positions the leading edge 406 to contact the

incoming air, A_{IN}, and such that the planar area of the tab 404 is substantially parallel to the flow path of the air, A_{IN}. In some embodiments, the tabs 404, 410 are substantially parallel to the simple flow path while in other embodiments, some, a majority, or all of the tabs 404, 410 are arranged substantially parallel (such as within about 5 to 10 degrees) to the local flow paths or streamlines. Tab 410 is shown to be generally square in shape but to include rounded shoulders 414 such that its leading edge is less likely to snag or catch debris in the incoming air, A_{IN}, that might clog the air flow channel between the fins and reduce heat transfer and/or increase pressure drop. Some of the tabs can be more semicircular in shape indicating that the shape of the tabs 404, 410 can vary on differing fins or within a single fin to practice the invention. The tabs can also bend at angles less than 50 degrees. In some situations, it may be advantageous to have tabs bend downward in the direction of gravity to facilitate water drainage from the fin surface that could result from rain, dew formation (as occurs in the case of an evaporator), and spray cooling enhancement.

Figure 5 is an enlarged view of air flow in a channel between a pair of tabbed fins of the present invention. As shown, the fin bodies 520, 521 have tabs 510 extending at a height, H_{TAB},, from the fin surfaces 528, 529. The fin bodies 520, 521 are separated by a fin separation distance, $D_{F.S.}$, and the tab height, H_{TAB}, is selected to be about one half of the fin separation distance, $D_{F.S.}$. The tab height may be varied and the illustrated embodiment is one useful embodiment. In typical finned tube heat exchangers, a boundary layer is formed on each fin surface 528, 529 as the incoming air, A_{IN}, flows through the fins 520, 521 from the fin body leading edge 530, 531 toward a fin body trailing edge 532, 533. As can be seen, the boundary layers extend outward from the fin surfaces 528, 529 creating an insulting layer that ends at about a midpoint between the fins 520, 521. Hence, it is desirable to have the tabs 510 extend at a height, H_{TAB}, that allows the fin material, i.e., the material in the tabs 510, to extend into (and in some cases, through) the outer portions of the main boundary layers and into the coolest

air flowing through the fins 520, 521. Again, the tabs 510 are illustrated as being substantially planar with a square shape with rounded or smoothed shoulders 514 to reduce snagging or filtering of debris in the air, A_{IN}. In one embodiment, the shoulders 514 have a radius of 0.009 inches but, of course, other larger and smaller radii may be implemented.

Figure 6 is a top view (or side view) of a fin or fin portion 600 fabricated according to the invention. As shown, when the fin 600 is in use, the cool air, A_{IN}, flows across a leading edge 611 of the fin 600 then across a first surface 610 of the fin body. The air then contacts a fin collar (and included tube not shown) 636 causing the creation of a wake or low pressure area 640 behind the collar 636 (and tube) and then, the air continues along the fin surface 610 until it passes over the trailing edge 612 where it is expelled as hotter air, A_{OUT}, of the heat exchanger in which the fin 600 is installed. The fin 600 includes numerous tabs that are formed by punching out sections of the fin body (but leaving the material attached on one edge, e.g., a proximal edge or a tab seam or connector) and bending or hinging the material upward away from the surface 610 or downward away from the opposite surface (not shown) of the fin 600. For example, a first tab 614 is bent downward relative to the surface 610 and the material removed from the body or surface 610 forms a tab opening or hole 616 adjacent the tab 614. A second tab 620 extends upward relative to the surface 610 at a substantially right angle with the removed material (i.e., the material retained in tab 620) creating a tab hole or opening 622 adjacent the tab 620. As shown, the tab holes 616, 622 (and corresponding tabs 614, 620) are substantially square in shape but other embodiments of fins of the invention may utilize other shapes.

As shown, the tabs are arranged generally in rows that extend substantially parallel to the leading and trailing edges 611, 612 of the fin 600. Note, that this particular configuration is not required but is useful for ease of tab pattern selection (such as relative to amount of surface area to be utilized), for ease of manufacturing, and

for assuring that tabs are positioned to achieve a desired sequentially offset arrangement. The offset feature of the invention can be seen by looking at the tabs in Rows 1-4 and particularly the four tabs of Rows 1-4 shown with the dashed line 630 that can be said to be corresponding or adjacent tabs in adjacent ones of the Rows 1-4. Row 1 can be thought of as the first row or most upstream row of tabs with Row 2 being the second row and immediately downstream row relative to Row 1 (or adjacent to Row 1). The tabs shown by line 630 in Rows 1 and 2 can be seen to be offset from each other. Likewise, the tab in element 630 in Row 3 is offset from the tab in element 630 in Row 2 (immediately upstream or in the adjacent row), and the tab in element 630 in Row 4 is offset from the tab in Row 3.

The amount of offset can vary to practice the invention, with the offset shown being one useful embodiment, e.g., the opening and tab in the downstream row is positioned substantially in the space between adjacent openings/tabs in the upstream row. Note, also, that the tabs in the element 630 are offset on a "diagonal" and this pattern is continued in several additional rows of tabs. However, other offset patterns may be utilized as long as corresponding tabs in adjacent rows are offset from each other. Preferably, the offset pattern is selected so as to provide a spacing between similarly positioned tabs, such as by skipping a number of rows before placing a tab in a similar position within a row (e.g., as shown in Figure 6, a pattern of 8 rows is used with 7 rows provided as a "spacer" before repeating a row pattern).

Figure 6 also illustrates that the pattern of tabs may be relatively uniform or, as shown, denser in areas of anticipated high flow of the cooling gas or air. Hence, as shown, there are fewer tabs placed behind the fin collar 636 (and other collars) where a wake is created by the collar 636 and, therefore, later installed tube (not shown). In this manner, a larger percentage of the tabs (and therefore, the area or material taken of the fin body 610) are placed in locations in between adjacent fins were heat transfer is most likely to occur effectively. Another way of stating this configuration is that the fin 600

is configured to have areas of higher porosity in areas upstream of tubes 636 than immediately downstream (e.g., areas of lower porosity in wake areas). Higher porosity or more tabs may also be provided between pairs of tubes to place more heat transfer surface in higher flow areas of channels between fin pairs. The spray pattern is also selected with due consideration to its impact on heat conduction within the fin.

In some embodiments, a subset (such as a small percentage such as 10 percent or less) of the tabs are purposely not aligned with the main air flow through the fin (i.e., not perpendicular to a leading edge of the fin) but are instead skewed or angled relative to the main or simple flow path or flow direction of the cooling gas so as to act as directional vanes. Typically, these directional vane tabs are positioned near (e.g., beside and/or slightly behind) the collars (and inserted tubes) of the fins to direct the air or gas flow into areas that otherwise would be starved for flow such as in the wake region behind the collar/tube. In other cases, the direction vane tabs may be further upstream to begin diversion of flow to the wake area prior to the tube collar and tube so that the flow redirection can be more gentle, i.e., less dramatic or turbulent. In one embodiment, at least some of the tabs near the collar 636 are angled to direct some air flow from the main flow path into the wake region 640. Preferably, the angle for the directional vane tabs is selected so as to avoid or minimize the creation of vortices behind these tabs so as to control increases in pressure drop, e.g., the angle may be less than about 5 degrees relative to the local flow paths or streamlines and the like. Unlike a delta winglet pair, the tabs in this embodiment gently direct the flow into the wake regions without causing turbulence. The reduction in wake size reduces form drag and overall pressure drop while at the same time providing better heat transfer coverage in the wake region behind the tubes/collar.

The introduction of the fins, although parallel to the simple flow path or local flow paths or streamlines, does alter the flow of the cooling gas relative to the fin, such as by increasing friction and by creating multiple thermal boundary layers within the gas

flow channel or passage. In some embodiments, patterns of the tabs are selected with the express purpose of gently redirecting flow of the cooling gas. For example, the pattern of the tabs are selected purposely to fold or direct more of the cooling gas into the wake areas and areas of low pressure or flow between two fins. In one case, the tabs are offset in diagonal patterns to gently orient (e.g., with minimal turbulence) toward the wake regions behind collars and tubes.

In still other embodiments of the invention (not shown), fins are fabricated according to the invention so as to generate at least some vortices in the cooling air or gas. In one vortex generating application, the tabs illustrated and described in the invention are utilized in combination with delta winglet pairs, such as near the tube collars or other areas of low flow. In this manner, the beneficial effects of the tabs of the present invention and of winglet pairs are combined to enhance heat transfer. The amount of pressure drop can be controlled by limiting the number of winglet pairs utilized, and/or this embodiment may be employed when a higher pressure drop is acceptable. In another vortex generating application, tabs that are sharply angled (such as over 5 degrees up to 90 degrees) relative to the main or local flow paths of the cooling gas are included on a fin. Typically, in these embodiments of the invention, the majority of tabs would remain aligned parallel with the main or local flow paths with a minority or small number of unaligned tabs being added in strategic locations, such as locations at which winglet pairs are often employed or other locations at which it is desirable to create turbulence.

Figure 7 illustrates a section of the fin 600. As shown, the tab 620 is formed by bending material upward from the surface 610 to form a hole or opening 622 in the fin 600. Tab 614 adjacent to tab 620 in the tab row is formed by bending material downward from surface 610 to form the hole or opening 616. The tabs 614, 620 are bent away from the fin (e.g., from surface 610 for fin 620) at a selected angle, i.e., a bend or punch angle, θ, that is typically selected to be about 90 degrees for ease of

manufacturing but similar or better heat transfer results may be achieved at smaller bend angles such as between 30 and 90 degrees. As discussed earlier, tabs bent downward or in the direction of gravity can enhance water drainage. The tabs 614, 620 have a tab height, H_{TAB}, that is shown to be about one half of the height of tube collar 636, which typically corresponds to the separation distance between adjacent fins. As discussed previously, the tab height, H_{TAB}, may be smaller or more preferably, larger such as 50 to 80 percent and in some tested cases, was about 67 percent to place a larger portion of the tab body or tab surface area within the cooler regions of air or gas flow in the channel between adjacent fins.

Figure 8 illustrates another embodiment of a fin 800 formed according to the presenting invention. The fin 800 may be useful in heat exchangers in which fins are utilized without tubes. The fin 800 also illustrates a more regular pattern of tabs and openings than in Figure 6 with the offset clearly being on a "diagonal," i.e., with tabs in adjacent rows being offset a selected distance from the immediately upstream row.

Figure 9 is a graph 900 of the exemplary flow paths of a gas at about 1 to 3 meters/second across the surface of a fin such as the fin 600 in Figure 6. As can be seen, most of the flow is along a path generally between the tubes or perpendicular to the leading edge of the fin. Note, that over much of the fin surface the flow is predominantly in one direction with the path lines changing only slightly with distance from the fin surface. Hence, except very near the tubes or in the wake regions, aligning the tabs on a fin along the main flow direction or simple flow path is useful for significantly increasing the heat transfer coefficient of a fin while avoiding or controlling introduction of drag or pressure drop. Such knowledge of the flow led to the design of the tabbed fins shown in Figures 2-8 in which the tabs are generally planar and aligned with the flow, i.e., positioned with their larger planar surfaces parallel to the main or simple flow path shown in Figure 9.

While machining costs and pressure drops may be increased, some embodiments of the invention can be fabricated with the tabs aligned more particularly with the flow path of gas corresponding to the location of the tab, e.g., alignment along the local flow path lines. In other words, the tabs may be arranged with many differing alignments to suit the flow in that particular region of the fin. It is expected, though, that the placement of the tabs in the fin would change the flow relative to the fin and it may take numerous iterations to "match" such a tab alignment to the flow. Additionally, flow patterns vary with other parameters such as gas or air velocity and the like, although laboratory tests have demonstrated that one tab pattern can work well over a wide range of air flow rates.

An exemplary tabbed-fin was fabricated according to the invention (similar to that shown in Figure 6) to allow the effectiveness of the above-described invention to be tested. To fabricate small test cores for transient testing, a punching tool was used in conjunction with a template from a CAD program that allowed concurrent punching (i.e., punching of tabs in both directions away from a fin body). Aluminum fin material was used to fabricate fins for testing, as is commonly used for finned-tube condensers with ½-in. tubes. An example of the template and fins used in the test are shown in Figure 2 as template 230 and fins 210.

Measurements at air flow of 3 m/s in early fin samples showed that the tabbed fins provided 68% more air-side heat transfer and had a 33% higher pressure drop than similar untabbed fins. For comparison purposes, small cores made up of advanced fin materials were also tested, e.g., wavy fins and louvered fins. These cores were of a different fin density, but it was believed useful to compare the performance of the tabbed fins to the plain fins for these other arrangements designed for enhanced heat transfer. Table 1 shows the test results for these cores and compares the results to the results obtained for the tabbed fins of the invention. Note that the wavy fins and louvered fins were in a test core containing only one row of tubes. Also, the tabbed fin

design had fins spaced at only 5 fins per inch (0.20 inch spacing) because that was the stock available. The tab-forming tool used produced tabs that are 0.050 inch on a side, so each tab extended only one-quarter way across the gap whereas it is believed that tabs that extend halfway or more across the gap perform better.

Enhancement Type	**Percent Change in Heat Transfer**	**Percent Change in Pressure Drop**
Wavy fins (one tube row)	-5	45
Louvered fins (one tube row)	56	136
Tabbed fins (3 tube rows)	68	33

Table 1. Percent changes in heat transfer and pressure drop for enhanced fins on ½-inch tubes compared to a reference of the same geometry but with plain fins (3 m/s approach velocity).

As discussed above, some embodiments of the invention comprise tabbed fins in which some or all of the tabs are arranged to be generally aligned with or to be substantially parallel (such as within 5 to 10 degrees or less) to the local flow path or local streamlines. Additionally, some embodiments may comprise tabs each with a body that is rectangular in shape so as to provide a more desirable aspect ratio to enhance heat transfer rates relative to the same overall porosity of a tabbed fin. With these design ideas in mind, a tabbed fin (or fin portion) 1000 is shown in Figures 10 and 11 that utilizes rectangular tabs extending from both sides of a fin body. The tabs are arranged at offset angles relative to the simple flow path of the fin 1000, i.e., a line drawn perpendicular to the leading edge 1020 of the fin body 1010, so as to align with the normal flow paths and avoid flow disruption and its associated pressure drop. Further, the tabs can be oriented at slight angles to the usual local path lines to help direct the flow into the stagnant wake regions behind the tubes.

As shown, the fin 1000 comprises a fin body 1010 with a plurality of tabs 1030 extending at bend angles of about 90 degrees from a first fin surface 1014 and a plurality of tabs 1040 extending in an opposite direction at bend angles of about 90 degrees from a second fin surface 1018. Generally, the tabs 1030, 1040 are alternated in each tab row, such as rows 1050, 1054, 1058 so as to create an overall tab pattern in which corresponding tabs or upstream/downstream tab pairs are offset or purposely not aligned along a line drawn perpendicular to the leading edge 1020 of the fin body 1010 so as to avoid shadowing downstream tabs with upstream tabs. In other words, heat transfer achieved in downstream tabs is enhanced by not placing the downstream tabs directly in wakes or vortices created by upstream tabs. Wherever possible, consistent with the above considerations, the hinged edge of a tab is located closest to the tube to minimize the heat conduction path length.

The fin 1000 further comprises tube collars 1026 for mating with heat transfer or fluid tubes (not shown). The tube collars 1026 define fin separation distances when the fin 1000 is mated with another fin in a heat exchanger (such as exchanger 100 in Figure 1). Each tab 1030, 1040 is generally rectangular (with or without a curved corner having a small radius to reduce collection of debris at sharp corners) with a tab body 1060 having a leading edge 1062 that initially contacts flowing gas or air. The tab body 1060 further has a top edge 1064 distal to the fin body 1010 and a trailing edge 1066. During use, it will be appreciated that higher heat transfer rates are achieved proximal to the leading edge 1062 but that the entire surface of the tab body 1060 contributes to transfer heat achieved by the tabs 1030, 1040.

Figure 11 illustrates more clearly that the tabs 1030, 1040 are fabricated with a bend angle of about 90 degrees measured relative to the fin body surfaces 1014, 1018 so as to extend substantially perpendicularly from the fin body 1010. Smaller or greater bend angles can be used but one of about 90 degrees is useful for extending the tabs 1030, 1040 further out into the cooler air flowing across the fin 1000 with less material

removal. As shown, the fin body 1010 has a porosity of about 25 percent that is achieved with fewer tabs 1030, 1040 than used in the fin arrangements of Figures 2-9, which used small, square tabs.

A simple flow line, $L_{Simple\ Flow}$, is illustrated as showing generally air flow across the fin 1000 during its use in a heat exchanger. As can be seen, the tabs 1030, 1040 are positioned at offset angles, θ_1, θ_2, relative to the simple flow path or line, $L_{Simple\ Flow}$, that enables the tabs 1030, 1040 to align with the usual local flow direction and, further, to direct the flow more uniformly over the fin surface and into the wake regions. As shown, the offset angles, θ_1, θ_2, relative to the simple flow line, $L_{Simple\ Flow}$, are about 10 degrees but the offset angles, θ_1, θ_2, may be larger (such as up to about 20 to 30 degrees) or smaller. The measurement of the offset angles, θ_1, θ_2, is provided as an absolute value or variance from the simple flow line, $L_{Simple\ Flow}$,, but could be also provided, as shown, as 10 degrees, 170 degrees, 190 degrees, and 350 degrees for the various tabs 1030, 1040 on the fin 1000. A streamline or line representing a local flow path, $L_{Streamline}$, is also shown in Figure 11 relative to the tabs 1030, 1040. The tabs 1030, 1040 are preferably aligned or positioned on the body 1010 such that the tabs 1030, 1040 are substantially parallel to (e.g., within about 5 degrees of) the local flow path or streamline, $L_{Streamline}$. Owing to their flow resistance and pattern and also, to their local angles, the tabs 1030, 1040 on the body 1010 act to direct flow, i.e., change the shape and location of local flow paths from those found in an untabbed or plain fin, to reduce wakes behind tube collars 1026 and tubes (not shown) while reducing drag or pressure drop increases associated with adding tabs 1030, 1040. The tab pattern of fin 1000 was obtained through some trial and error because the introduction of tabs 1030, 1040 altered the shape and location of local flow paths, $L_{Streamline}$, and several iterations were required to obtain the alignment of the tabs 1030, 1040 with the resulting streamlines, $L_{Streamline}$. Those skilled in the art will understand that the resulting streamlines, $L_{Streamline}$, may vary with the shape, size, and number of the tabs 1030, 1040, the

location, shape, and number of the tube collars 1026, the speed of the air or gas flow across the fin 1000, and other parameters. However, it is believed that the teaching provided in this description can readily be extended to many other tube and fin configurations.

To more fully describe the invention, it may be useful to more fully describe one fabricated embodiment of the fin 1000 of Figures 10 and 11 and provide typical test results. One particular embodiment or implementation of the fin 1000 was configured for a finned tube heat exchanger (such as, but not limited to, the heat exchanger 100 of Figure 1) that has 8 fins per inch (FPI). However, the fins 1000 can readily be used in different FPI exchangers such as 10 FPI exchangers. In the particular 8 FPI embodiment of the fin 1000, the tab length as measured along the top edges 1064 was 0.120 inches. The fin separation in the heat exchanger (not shown) was about 0.115 inches and the tabs 1030, 1040 were designed to have a tab height as measured along the leading and trailing edges 1062, 1066 of tab body 1060 of about one half the fin spacing or about 0.0575 inches. However, as discussed earlier, it is preferable in some cases to have a tab height that is greater to place more area of the tab surface in cooler portions of air flow, and in these cases, the tab height may be about two thirds or more of the fin separation or about 0.0771 inches in the 8 FPI embodiment. The corners are rounded with a radius of about 0.009 inches. The tab edges, such as edges 1062, 1066 are not perpendicular as in a typical rectangle but are instead angled inward at about 5 degrees giving the tab body 1064 a trapezoidal shape or substantially rectangular shape. The offset angles, , θ_1, θ_2, were set at about 10 degrees in this particular embodiment of the fin 1000.

To model and understand flow of air through channels defined by two adjacent, tabbed fins, a bubbler device was utilized by the inventors. In Figure 12, a drawing 1200 representing flow visualization photographs is provided for a plain or untabbed fin 1210. The bubbler device creates numerous bubbles that flow in fluid passing over the fin 1210 and the bubbles help to define or allow visualization of local flow paths or

streamlines, identified as element 1218. Air coming in, Air_{IN}, in a plain fin would pass over leading edge 1212 of the fin and generally flow in streamlines 1218 that are generally perpendicular to the front edge 1212 until the streamlines 1218 pass between tube collars 1220 and out of the channel as hotter air, Air_{OUT}. At this point, the local flow paths are directed inward away from and around the tube collars 1220 and create wakes 1230, which have are relatively long with a wake length, $L_{WAKE\ 1}$, of about 1 to 1.5 inches with ½-inch tube collars or tubes at 3 meter/second flow. The streamlines 1218 are also disrupted by inner tube collars 1230 and another wake 1240 is created with stagnant, separated flow and again the wake 1240 has a length, $L_{WAKE\ 2}$, that may be about 1 to 1.5 inches. The creation of wakes 1230, 1240 is important because in the wakes 1230, 1240 heat transfer with the tube/tube collars and/or fin 1210 is dramatically reduced when compared with other areas of the fin 1210 or when compared with the front of the tube/tube collars because of the stagnant flow in these regions. An important design goal associated with implementing tabbed fins, therefore, is to reduce the size of the wakes 1230, 1240 or to at least increase heat transfer in these areas of the fin 1210.

Figure 13 illustrates with flow visualization 1300 the effect the tabs 1314 have on air flowing across, Air_{IN}, a tabbed fin 1310 before being discharged as Air_{OUT}. The tabbed fin 1310 is configured with square tabs 1314, such as 0.050-inch by 0.050-inch tabs, that are aligned substantially perpendicular to the front or leading edge 1312 of the fin 1310 so as to be generally parallel to the simple flow path of the air relative to the fin 1310. The pattern used in fin 1410 is similar to that shown in Figures 2 and 6. The local streamlines 1318 are shown less compressed between the tube collars/tubes 1320 as the flow channels and pressure drop caused by the tabs 1314 act to distribute the flow more uniformly and redirect flow forcing at least a portion to flow more directly behind the tube/tube collars 1320. As a result, the side wakes 1330 have a length, $L_{WAKE\ 1}$, that is shorter, such as about 0.5 to 1 inch rather than 1 to 1.5 inches. Similarly, the pattern

of tabs 1314 near the inner tube/tube collar results in a wake 1340 that has a wake length, $L_{WAKE\ 2}$, that is significantly smaller.

Figure 14 illustrates with flow visualization 1400 the usefulness of a tabbed fin 1410 comprising at least some tabs 1414 that are bent or positioned at an offset angle relative to the simple flow path. The pattern of the tabs 1414 is the same as in the fin 1310 of Figure 13 except that a portion of the tabs 1414 have been bent at an offset angle between about 10 and about 15 degrees relative to the simple flow path. In this manner, the incoming air, Air_{IN}, is allowed to flow in local streamlines 1418 but is also directed to flow into the wakes 1430 and 1440 so as to create wakes with lengths, $L_{WAKE\ 1}$ and $L_{WAKE\ 2}$, that are significantly shorter than the wakes of a plain fin and even shorter than tabbed fins for which the tabs are not positioned at an offset angle (or the offset angle is zero). As a result, the air flowing out, Air_{OUT}, is able to absorb more heat from contact with the tabs 1414 and the tube/tube collars 1420 and fin surfaces. As will be understood, numerous patterns of the tabs 1414 can be used to reduce wake size by directing the air flow but an important consideration is the trade off between redirecting air flow to reduce wake size and the undesirable increase in drag and pressure drop that can occur with tabs with too large an offset angle. Generally, tabbed fins according to the invention utilize offset angles that are relatively small, such as less than 15 degrees, so that the tabs remain substantially parallel with the simple flow path and present a relatively small contact profile with the flowing gas or air.

Testing of a set of fins 1000 shown in Figure 10 and 11 was performed by the inventors in a 8 FPI configuration and in a 10 FPI configuration with comparison provided to plain fins in the same configurations. The results are provided in the graphs in Figures 15-20. In the 8 FPI configuration, the fin separation distance or gap was 0.115 inches for the plain and tabbed fins and was 0.09 inches in the 10 FPI configuration for both types of fins. The tabs had a height of 0.0575 inches which resulted in the 8 FPI configuration testing tabs extending 50 percent of the fin separation

distance while in the 10 FPI the fins extended to approximately two thirds of the gap or more particularly about 64 percent of the fin separation distance. This resulted with fins in adjacent fins actually overlapping by about 0.025 inches, but the tabs were arranged so as not to contact or abut when the heat exchanger or test assembly was fabricated. The porosity of the tabbed fins was approximately 25 percent.

Turning to Figure 15, a graph 1500 is provided of the results of testing a plain, aluminum fin at varying air flow velocities. Determined heat transfer coefficients and isothermal pressure drops are plotted in the graph versus varying air flow velocities, which are typical of conventional air cooled condensers. Line 1510 represents values of the heat transfer coefficient for the plain fin while line 1520 represents values of the pressure drop in the plain fin, 8 FPI configuration. As shown, in the range of 2 to 3 m/s face velocity the heat transfer coefficient ranges from about 38 to 50 W/m^2K while the pressure drop ranges from about 22 to 43 Pa.

Figure 16 provides a graph 1600 similar to graph 1500 showing values of the heat transfer coefficient for a tabbed fin according to the invention (see Figures 10 and 11) with line 1610 and values of pressure drop with line 1620 at varying face velocities. As can be seen, there is a significant improvement in the heat transfer coefficient for the tabbed fin compared with the plain fin. Specifically, in the same face velocity range of 2 to 3 m/s, the heat transfer coefficient ranges from about 65 to about 85 W/m^2K. This represents about a 70 percent increase in the heat transfer coefficient for the tabbed fin relative to the plain fin of similar size, material, and thickness. The "cost" of this added efficiency or effectiveness of the tabbed fin is an increase in pressure drop. However, as shown, the pressure drop for the tabbed fin in the 2 to 3 m/s face velocity range is only about 31 to 62 Pa which represents a relatively low increase in pressure drop of about 50 percent that would likely be an acceptable tradeoff for the significant increase in the fins heat transfer rate and in effectiveness of heat exchangers incorporating such tabbed fins.

Figure 17 illustrates a graph 1700 of ratios comparing design parameter values of the tabbed fin with the plain or original fin plotted against the varying face velocities used during the testing of the fins. As shown, line 1710 represents values of a dimensionless heat transfer coefficient called the Colburn *j* factor for the tabbed fin compared with the plain fin. Line 1720 represents values of the friction factor (i.e., a dimensionless pressure drop for internal flow) for the tabbed fin over the valued for the plain fin. As can be seen from these two ratios, the heat transfer coefficient is consistently greater than for the plain fin but so is the pressure drop. Line 1730 represents values of a ratio of the above two ratios. This ratio is significant in that the ratio is greater than one over the various face velocities indicating that the increase in heat transfer coefficient values is always greater than the corresponding increase in pressure drop. Finally, line 1740 illustrates values for the design equation of $(J/J_O)/(F/F_O)$^(1/3) that is utilized by heat transfer designers in determining whether a design change, such as tabbed fins, can provide more heat transfer at the same fan power.

Figures 18-20 provide results of testing similar to those provided in Figures 15-17 but for a fin configuration of 10 FPI. Figure 18 illustrates a graph 1800 of the results of a plain fin configured at 10 FPI with line 1810 representing heat transfer coefficient values determined for the plain fin at various face velocities. When face velocities vary range from 2 to 3 m/s, the heat transfer coefficient ranges from about 40 to 48 W/m^2K. Line 1820 represents pressure drop values for the varying face velocities, and in the range of 2 to 3 m/s face velocity ranges from about 28 to 55 Pa.

Figure 19 provides a graph 1900 showing similar test results for a fin tabbed as shown in Figures 10 and 11 with a porosity of 25 percent and placed in a 10 FPI configuration. As shown with line 1910, the heat transfer coefficient is greatly improved ranging from about 80 to about 98 W/m^2K when the face velocity ranges from 2 to 3 m/s. This represents an increase in the heat transfer coefficient for the tabbed fin

relative to the plain fin of about 100 percent. Line 1920 represents the corresponding pressure drop, which in the 2 to 3 m/s face velocity range has a value of about 45 Pa to about 85 Pa. The "cost" is then an increase in pressure drop for the tabbed fin of about 55 to 60 percent, which as with the 8 FPI configuration would be an acceptable price to pay for such a large enhancement of heat transfer effectiveness for the air side fins and for heat exchangers incorporating such tabbed fins. Figure 20 illustrates a graph 2000 of design ratios similar to graph 1700 of Figure 17 with lines 2010, 2020, 2030, and 2040 representing values of heat transfer design ratios. As with the 8 FPI design, the tabbed fin testing in a 10 FPI configuration shows through the values of the design ratios in graph 2000 that the tabbed fin concept of the present invention, and particularly, the pattern of the fins in Figures 10 and 11, has merit and should be considered further for inclusion in heat exchangers, and especially, for inclusion in air cooled heat exchangers in which fan power is a limited resource.

As can be seen, the tabs may be selected to have a height as measured from the fin body that varies significantly to practice the invention, but that will typically be selected to be about the fin separation distance or less to avoid problems in fabricating the heat exchanger. More typically, the tab height is selected to remove adequate material from the fin body to have the tab body extend out into the cooler flowing air. Generally, testing has shown that it is desirable for the edge of the tab distal to the fin body to extend beyond the top of the boundary layer so as to place a significant portion of the tab body surface area in the very coolest air. With this in mind, most embodiments of the invention utilize a tab height in the range of about 25 to 75 percent of the gap or fin separation distance. For an 8 FPI exchanger, the gap is about 0.115 inches and the tab height is selected from the range of about 0.029 inches to about 0.087 inches. More typically, the tab height is selected from the range of about 40 to 67 percent of the gap. In these cases, the tab height is in the range of about 0.0460 inches to about 0.077 inches for the 8 FPI exchanger. In a 10 FPI heat exchanger, the gap is

smaller at about 0.09 inches and hence, the tab heights would be selected from the overall range of about 0.022 inches to about 0.0675 inches while the narrower range is about 0.036 inches to about 0.06 inches.

While generally the tabs or a majority of the tabs are aligned so as to control pressure drop, it may be useful in some embodiments of the invention to have a subset of the tabs purposely arranged or configured to generate turbulence. These turbulence generating tabs may, for example, be arranged with offset angles greater than 10 degrees, e.g., at or near 90 degrees but often at lower offsets, and may be located on the fin in select areas to create higher heat transfer in otherwise low heat transfer areas (such as behind the collars and/or tubes) or may be interspersed among the other tabs. The number of tabs in the subset relative to the other tabs may vary widely to practice the invention and will typically be driven by allowable pressure drop for a heat exchanger application.

In other embodiments (not shown), a transition to turbulence can be promoted by the configuration of the tabs and base fin by manipulating surface roughness or texture. Generally, the tabs and base fin have low surface roughness, e.g., are smooth metal. In some embodiments, though, surface roughness of the base metal is increased to a desired amount to cause adjacent flow to begin to transition to turbulent flow. The surface roughness can be thought of as a surface treatment and may include (or be replaced by) dimples or other surface treatments that alter the surface texture from smooth to a level of roughness that promotes turbulent flow. In one embodiment, the surface treatment is applied only to tabs (or portions of each tab or a subset of the tabs) while in others the treatment is applied only to the fin and/or tubes. In other cases, the surface treatment may be applied to all of these components or any combination.

The above disclosure sets forth a number of embodiments of the present invention. Other arrangements or embodiments, not precisely set forth, could be

practiced under the teachings of the present invention and as set forth in the following claims. Particularly, the use of tabs aligned and configured as discussed in the above description is readily applicable with fin arrangements other than parallel plate arrangements. For example, air-cooled condensers are often configured with tubes upon which fin material (such as aluminum foil) is helically wound. Such condensers may readily incorporate tabbed fins to enhance heat transfer, such as by punching the fin material prior to winding the material onto the tube. The tabs illustrated in the figures are generally planar with a rectangular cross-section when viewed from the leading edge. Other tab cross sections can readily be envisioned and are considered within the breadth of the above disclosure. For example, tabs with an upside down "L" cross-section can be substituted for the illustrated tabs and may be useful for placing greater tab surface area in the cooler air flow while not unacceptably increasing drag and/or manufacturing costs. Other tab cross sections include a stepped cross section, a wavy or serpentine cross section, an L-bend, a U-bend, and the like. Fin bodies can be fabricated with tabs extending from both sides by forming a composite fin from two fins with tabs extending from one side and their planar surfaces abutting.